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LUATION OF LIGHTWEIGHT, COMPOSITE, IMPACT-
STANT TAIL ROTOR DRIVE SHAFTING FOR HELICOPTERS

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May 1979

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Three contractors, Whittaker Corporation, Lockheed-California Company, and Fiber Science Incorporated, each designed and fabricated five samples of a lightweight, composite, impact-resistant tail rotor drive shaft. The shaft specimens were proof tested by the contractors, and ATL performed evaluation testing. Test results indicated that both sandwich-wall and wet-filament-wound drive shafts exceeded stated design requirements, and that production of high quality composite drive shafts is feasible. The majority of test failures occurred in shaft end fittings, indicative that additional research and development effort is needed in end fitting design.

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INTRODUCTION

The Applied Technology Laboratory (ATL) has conducted two significant programs to investigate the low energy impact damage tolerance of graphite. The first study was performed by McDonnell Douglas Aeronautics Company. The objective of the study was to obtain an understanding of the graphite brittleness phenomenon and some of the variables that affect the phenomenon. The program was completed in April 1975, and a final report was published.[1]

The second program, which dealt directly with graphite tail rotor drive shafts, was a combined effort on the part of Whittaker Corporation Research and Development Division (WRD)[2], Lockheed-California Company[3], and Fiber Science, Incorporated (FSI)[4]. Each contractor was required to design, fabricate, and test two shaft assemblies to demonstrate that they met the Army critical speed and torsional strength requirements. If the shafts met the requirements, the contractor was to fabricate five assemblies for delivery to ATL for evaluation testing. This technical note presents the results of the joint effort.

---

[1] Greszczuk, L. B., and Chao, H., *Investigation of Brittle Fractures in Graphite-Epoxy Composites Subjected to Impact*, McDonnell Douglas Aeronautics Company; USAAMRDL Technical Report 75-15, Eustis Directorate, U.S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia, May 1975, AD A012269.

[2] Hilzinger, J. E., and Berg, K. R., *Design and Fabrication of Lightweight Impact-Resistant Drive Shafting (Single Wall)*, Final Report MJO 3044, Whittaker Corporation, Research and Development Division, San Diego, California, January 1975.

[3] Griffin, C. F., *Design and Fabrication of Lightweight Impact-Resistant Drive Shafting*, LR 26708, Lockheed-California Company, Burbank, California, 14 November 1974.

[4] Abildskov, D. P., Ashton, L. P., and Yao, S., *Development Program for Filament Wound Drive Shafting*, Final Report, Fiber Science, Inc., Gardena, California, September 1974.

DESIGN REQUIREMENTS

The design requirements specified by the Government for this program are presented in Table 1. The requirements are not based specifically on a particular helicopter, but shaft size and loads are typical of those on a 14,000- to 16,000-pound helicopter.

TABLE 1. DRIVE SHAFT DESIGN REQUIREMENTS

Maximum outside diameter	4.25 in.
Minimum length (excluding end fittings)	42 in.
Ultimate torsional load	11,000 in.-lb
Fatigue torsional load	4600 ± 550 in.-lb
Operating speed	7000 rpm
Minimum critical speed	8000 rpm

The shafts must be capable of meeting the design requirements after sustaining either a 12 ft-lb ball drop, a .50 caliber AP impact at 2900 fps, or a fully tumbled .30 caliber AP impact at 2700 fps. The shaft should be designed with end fittings that are either integral with or attached to the tube ends. The end fittings must be at least of equivalent strength to the basic tube; i.e., any failure, whether in torsion or fatigue, must occur in the tube. Detailed stress analysis techniques are to be developed and applied to the shaft design.

DESIGN – GRAPHITE TAPE WOUND

Preliminary Design

Prior to the finalization of the design of the graphite tape-wound drive shafts, a preliminary design analysis was conducted and trade-offs were defined based on the design requirements in Table 1. The analysis resulted in a drive shaft (torque tube) that met those requirements. Refinements of the preliminary analysis were then used to predict the behavior of two full-scale drive shaft assemblies (torque tubes 1 and 2). These tubes had the properties listed in Table 2.

TABLE 2. TORQUE TUBE PROPERTIES DATA

	Tube 1	Tube 2
Overall ..ngth, in. (including end doublers)	55.0	55.0
Net length of free tube between end doublers, in.	45.4	45.4
Inside diameter, in.	3.7535	3.7535
Wall thickness, in.	0.0385/0.040	0.0375/0.0385
Average ply thickness, mil	6.54	6.33
Resin content, %	40.9	40.7
Specific gravity, gm/cc	1.495	1.497
Fiber volume, %	51.0 ± 0.5	51.2 ± 0.5
Outside diameter with skin, in.	4.234	-
Tube weight, bare, lb (including end doublers)	1.66	1.66
With end fittings	2.76	2.76
With skin and core	3.03	-
Maximum tested speed, rpm	8170	8130
Tube weight (uniform section) including skin and core, lb/ft	0.384	0.384
Maximum torsion load, in.-lb	15,400	15,932
Maximum torsion strain, in./in.	0.0106	0.0105

Based on the actual properties of the tubes as shown in Table 2, the following properties can be predicted:

Ultimate shear strength. The ultimate strength is based on design allowables for a nominal laminate of 60-percent fiber volume fraction (V_f). The allowable shear strengths are based on the ± 45° layers, which are Thornel 300. In the Air Force Design Guide[5]

[5] *Advanced Composites Design Guide*, Third Edition, U.S. Air Force Flight Dynamics Laboratory, Wright-Patterson AFB, Ohio, September 1976.

the nominal laminate shear strength is given as 21,600 psi. The predicted ultimate strength is within 2.5 percent of test results. In addition, the ultimate strength of the two test shafts differed by only ± 1.1 percent (see Table 3).

TABLE 3. ULTIMATE STRENGTH

	Tested F_{su} (avg) (psi)	V_f (avg) (%)	Predicted Allowable F_{su}* (psi)
Small evaluation tubes	20,000	59.2	21,300
Full-scale tubes			
1	17,900	51.0	18,100
2	17,500	51.2	18,200

*Air Force Design Guide (Corrected for V_f)

Note: These data points are compared in Figure 1.

Torsional Buckling Strength. The torsional buckling strength of a tube is a function of its length and its cross-sectional properties. Based on the data in Table 2 and using an orthotropic computer program, the buckling shear stress is calculated. Since the effective tube length is approximately 50 in., the torsional buckling strength is calculated for lengths from 45 in. to 55 in. The tube becomes buckling critical for a length between 60 and 65 in. The actual tube wall thickness varies between 0.0375 in. and 0.040 in. The buckling torque is therefore a band for this range of thickness (see Figure 1).

Torsional Stiffness. The torsional modulus of the tubes is predicted to be 1.99 x 10^6 psi for V_f = 60 percent, based on the Air Force Design Guide. The average of the small tube test data was 1.9 x 10^6 psi for a V_f (average) of 59.2 percent. A summary of the shear modulus of the small tubes and the full-scale test shafts with a comparison with predicted values is shown in Table 4.

TABLE 4. SHEAR MODULUS

	Predicted G (psi x 10^6)	V_f (%)	Tested G (psi x 10^6)
Small tubes	1.96	59.1	1.90 (avg)
Tube 1	1.70	51.0	1.72
Tube 2	1.71	51.2	1.60

Ultimate Shear Strain (See Figure 2). The ultimate shear strain, although not a design requirement, is of value in monitoring test results and repeatability. Since ultimate strain is particularly sensitive to changes in failure mode, local imperfection or irregularities, and stress concentrations, it generally has a greater scatter.

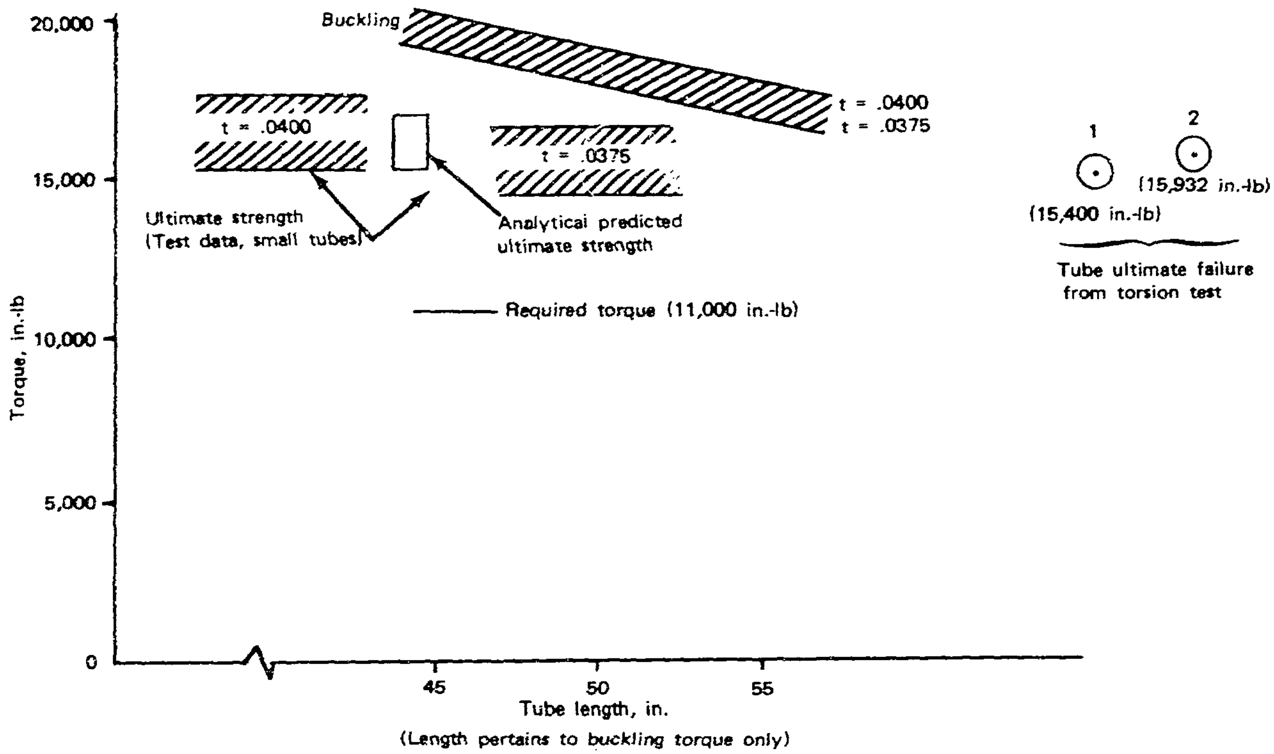


Figure 1. Torque tube strength.

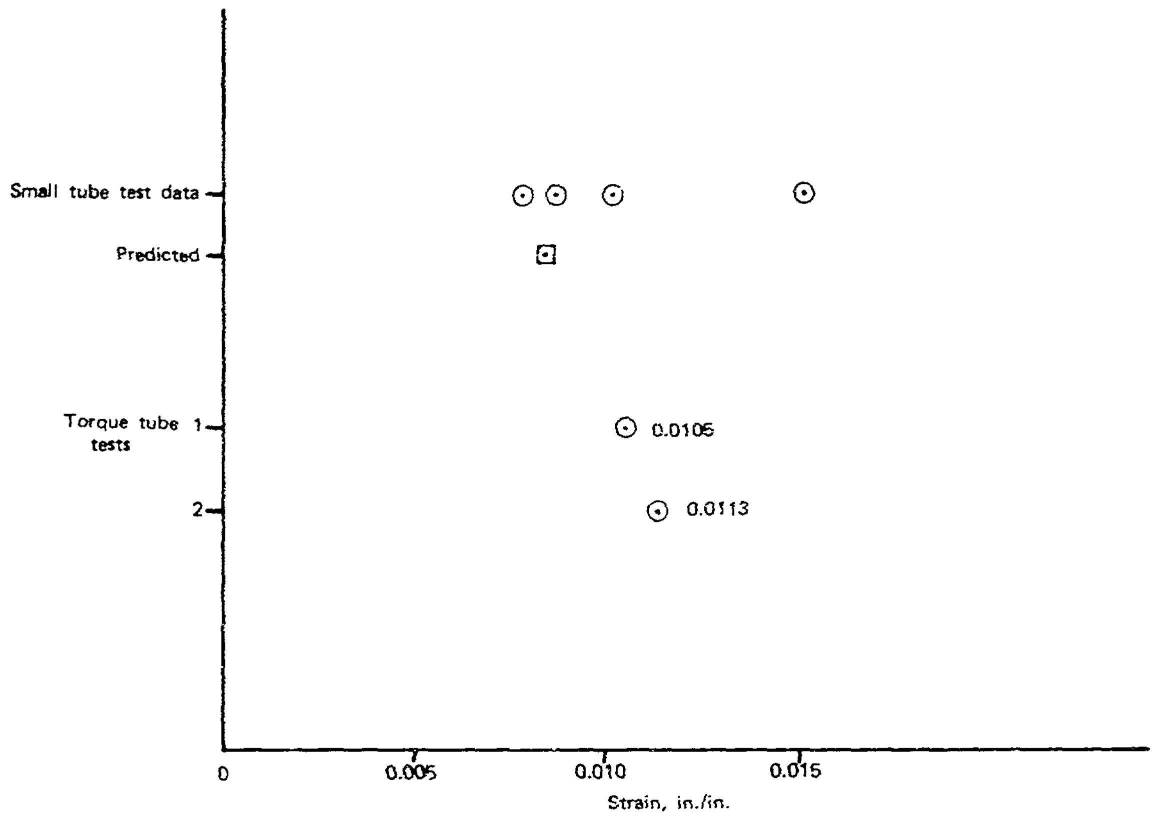


Figure 2. Torque tube strain.

Critical Speed (see Figure 3). The critical speed is calculated directly from a consideration of the longitudinal stiffness and length of the shaft. Although the shafts were not rotated to demonstrate their actual critical speed, sufficient rotational speed was achieved to demonstrate no critical spped below 8000 rpm.

Final Design

The parameters described in the preliminary analysis for optimized design were essentially included in the final drive shaft design. The six-ply configuration investigated (Figure 4) included the use of Thornel 300 for the $\pm 45^{\circ}$ ply orientations and Thornel 75S for the other plies. The small-scale tube fabrication, however, produced a wall thickness of 0.038 in., almost 0.001 in. thicker per average ply than the 0.0055 in. used in the analysis. Consequently, the inside tube diameter was fixed at 3.75 in. to permit the anticipated composite thickness and still allow sufficient depth (0.2 in.) for an aluminum Flex-Core covering. Additional allowance was included in the design envelope to permit the use of an outer shrink tube covering up to 0.010 inch thick. Ultimately, a 2 lb/ft^3 foam was added to the honeycomb to enhance the low-velocity impact protection. A second 0.005-in. polyvinyl chloride (PVC) cover was also added for the same purpose.

Completing the torque tube assembly are two lightweight aluminum end fittings bonded at mating tapered surfaces. Composite doublers at $\pm 45^{\circ}$ are then added to reinforce those areas over the bonded joint. The final length of the test drive shaft assemblies is 56.6 in. from fitting to fitting.

In order to facilitate tube testing, aluminum adapters were designed to permit direct attachment to the Army's torsional test fixture.

The overall design emphasis is directed toward high performance and impact protection with an efficient low weight structure.

Tooling and Fixture Design

The tooling requirements for the program were relatively modest and consisted of a shaft fabrication layup mandrel (Figure 5), a conical shaping tool (Figure 6), and a locating fixture for torque tube end fittings (Figure 7).

The layup mandrel is 60 in. long and extremely uniform over the entire polished length. It has provision for bearing mounting and is driven by a variable-speed drive during actual tube manufacture. The shaping tool is self-centering and was designed to provide a slight taper internally at each end of the composite tube to match the end fitting configuration. After installation of the end fittings into the graphite/epoxy tube, a locating fixture was used to provide precision alignment of the assembly during cure.

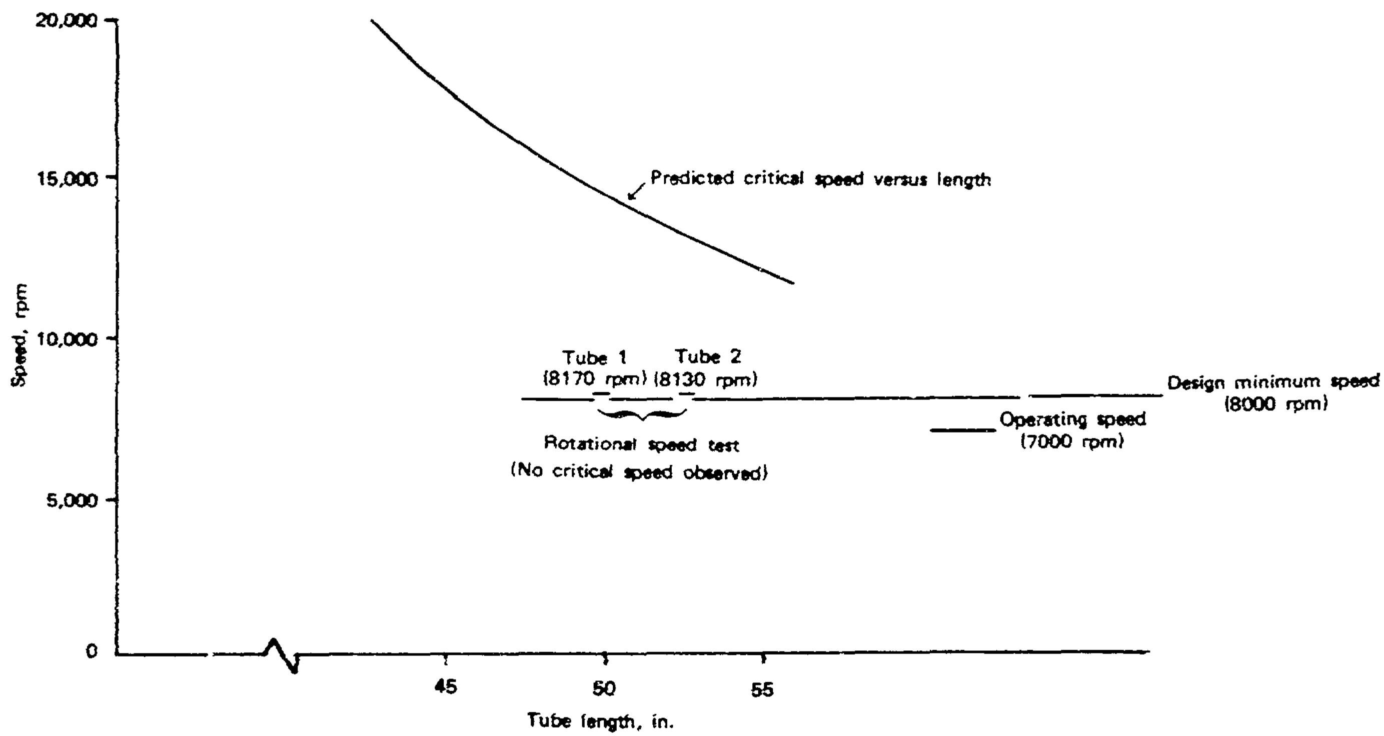


Figure 3. Torque tube critical speed.

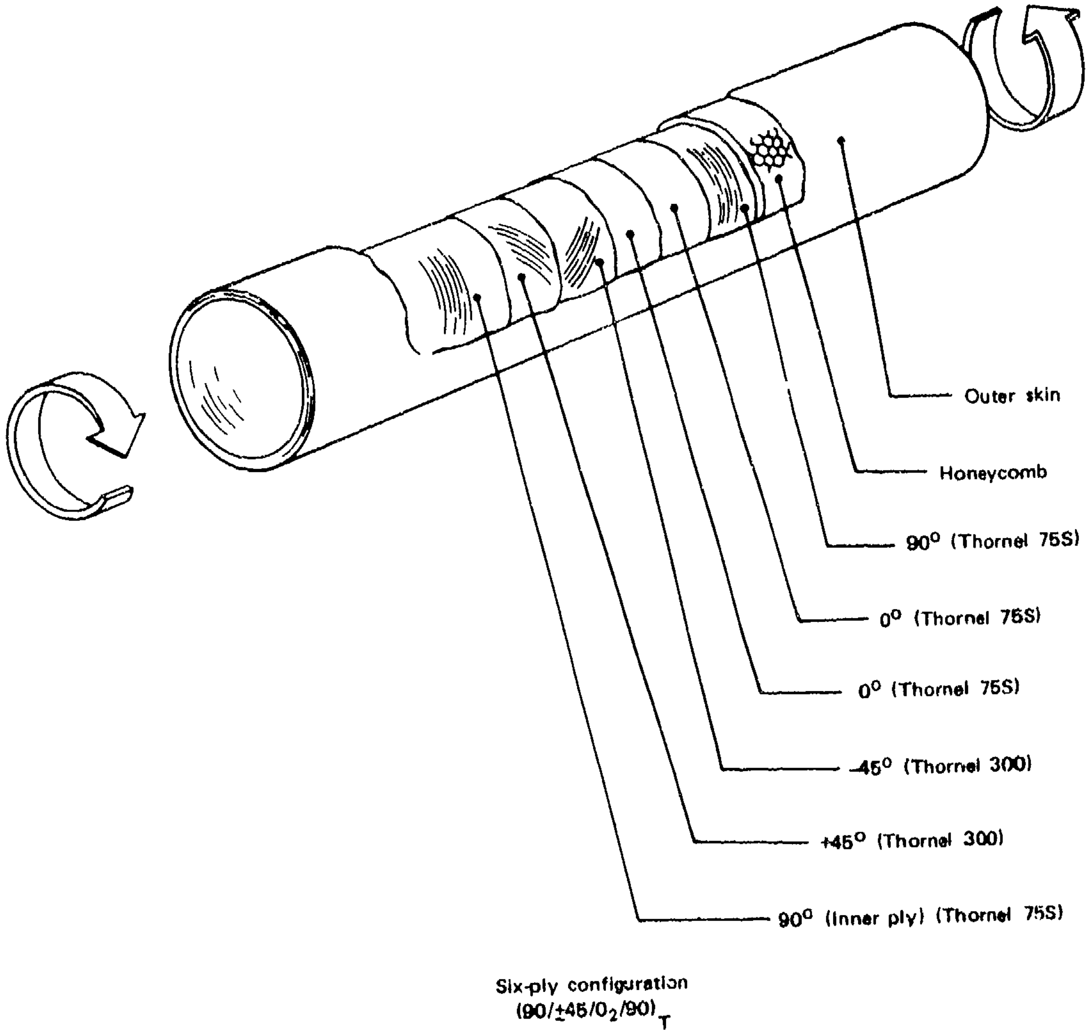


Figure 4. Tube layup.

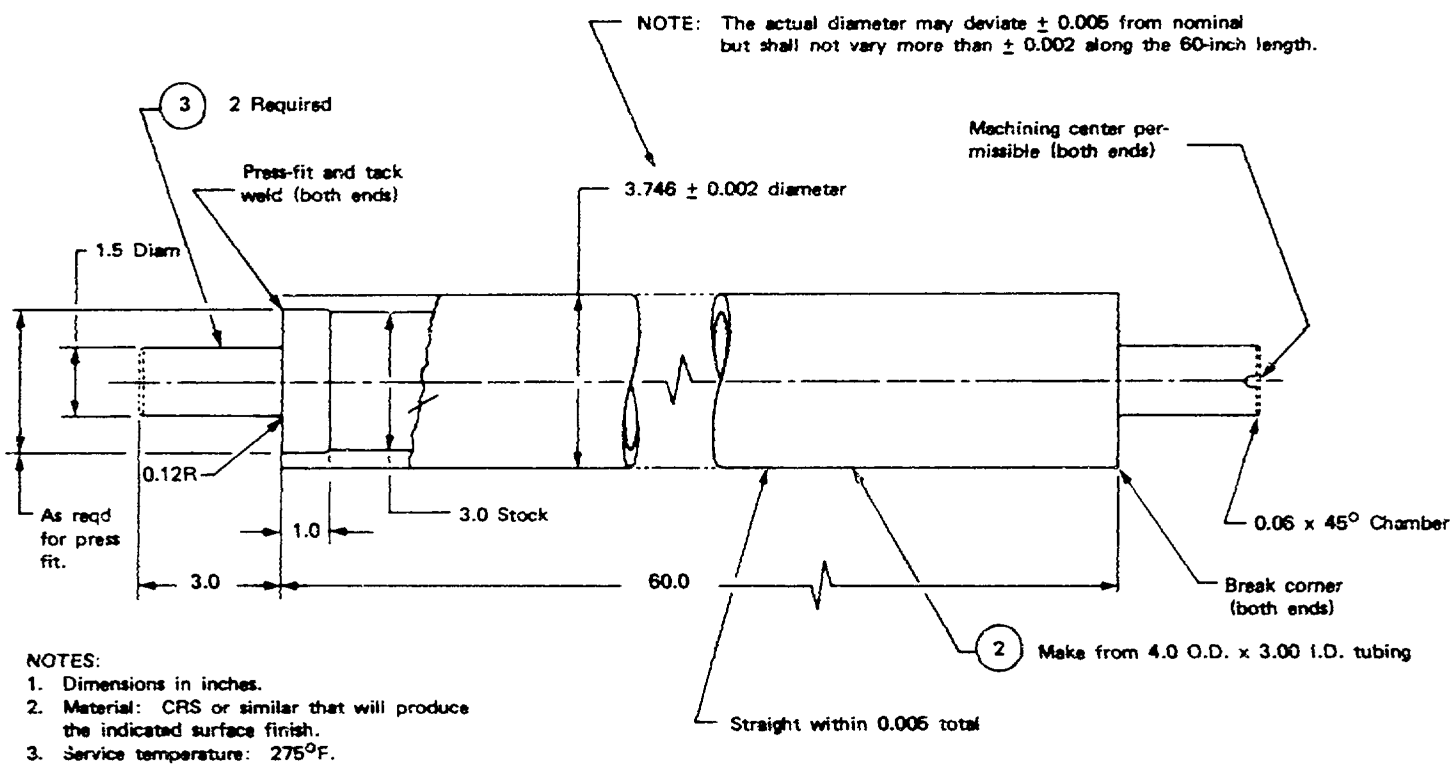
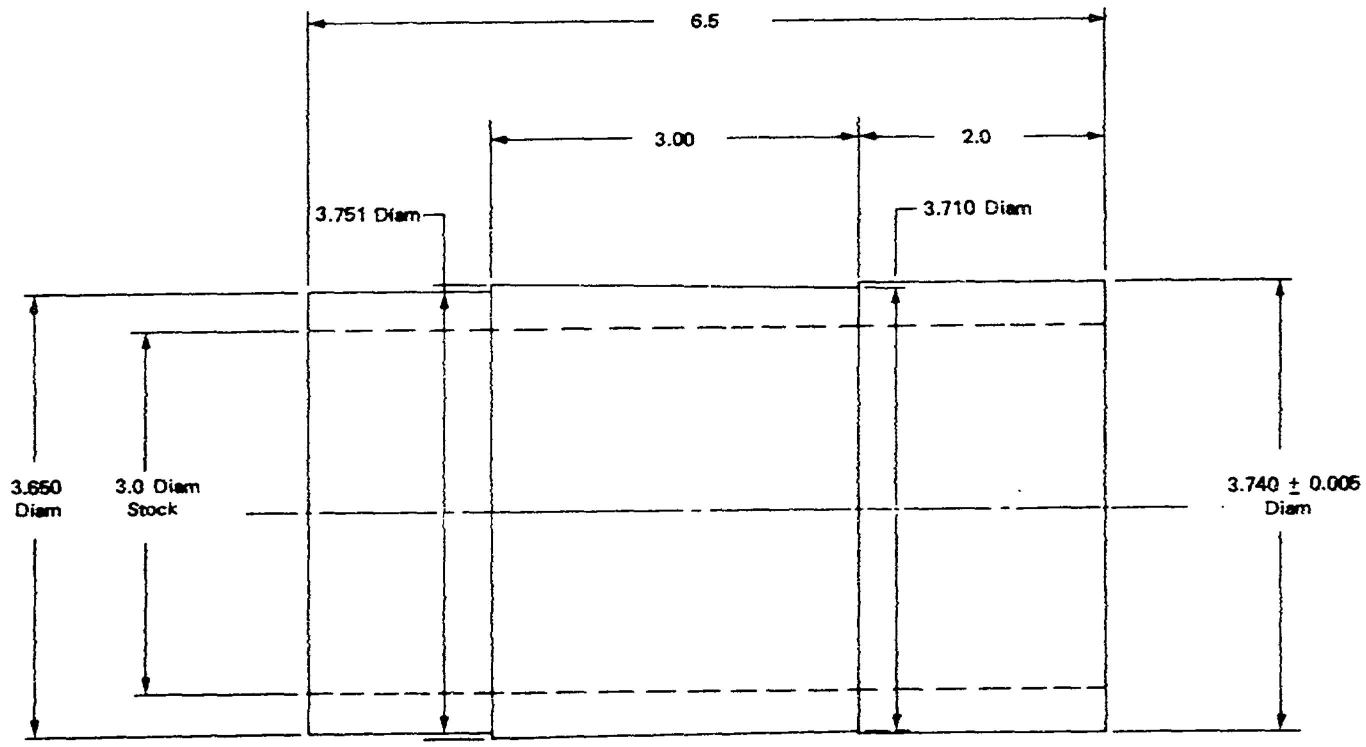
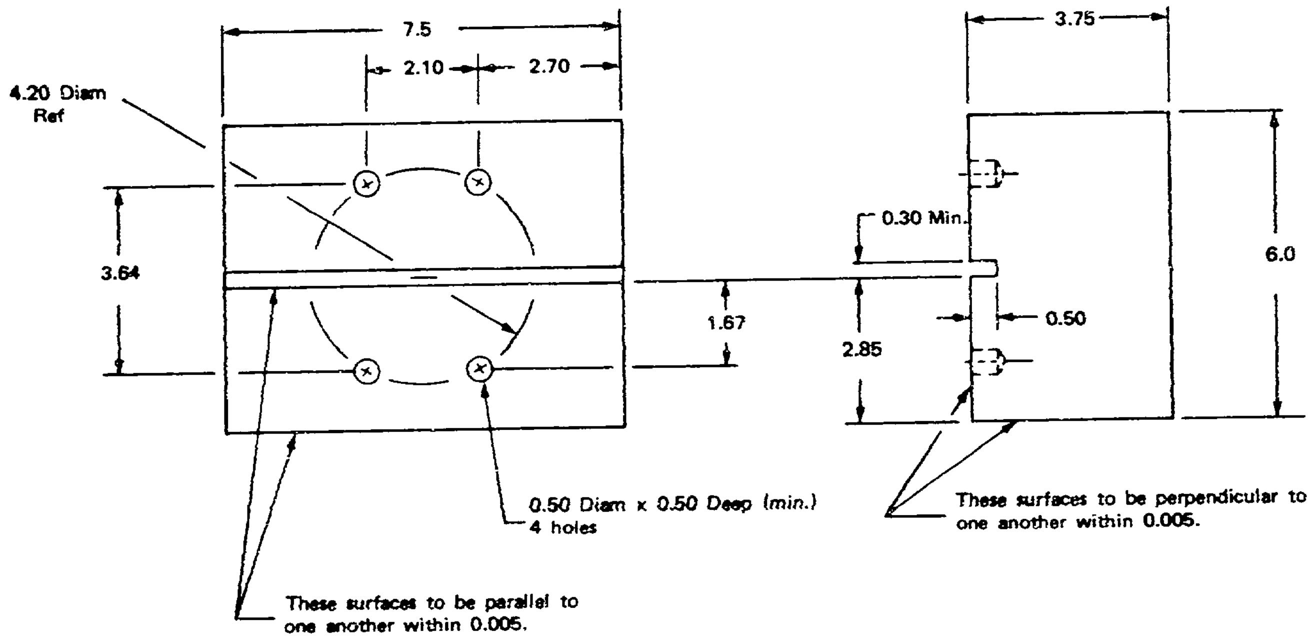


Figure 5. Mandrel - Graphite torque tube.



NOTES:
1. Dimensions in inches.
2. Make from 4.0 O.D. x 3.0 I.D. tubing
Material: 6061-T6 or 2024-T4 *aluminum* (Commercial grade)

Figure 6. Shaping tool, conical - torque tube.



NOTES:
1. Dimensions in inches.
2. Material: 2024 or 6061 aluminum

Figure 7. Locating fixture, end fitting - torque tube.

FABRICATION

Prepreg Fabrication

The preliminary analysis of the tube design included a target ply thickness of 0.005 in. In order to achieve this condition with both types of graphite reinforcements, a number of flat laminates were fabricated from prepreg containing a range of fiber spacings. As seen in Figure 8, Thornel 75S required 60 tows/in. to produce the 0.0055-in. average ply thickness.

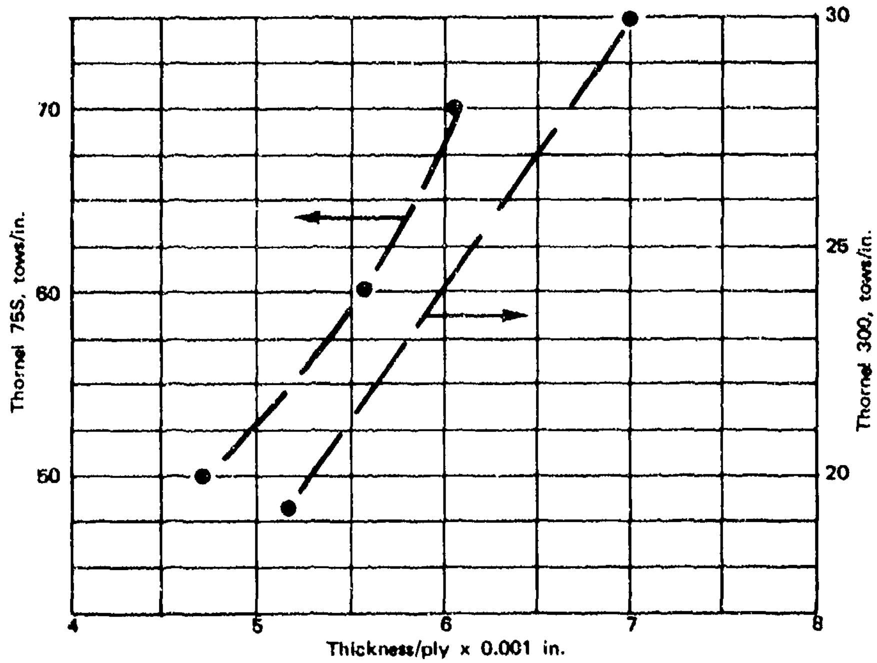


Figure 8. Fiber spacing versus ply thickness for Thornel 75S and Thornel 300.

However, Thornel 300 achieved the same ply thickness at a 21-tow/in. spacing. These fiber distributions were maintained throughout the program in the epoxy prepreg system, with a nominal resin content of 40 percent.

Vendor reports of fiber properties indicate far more variance in the ultimate tensile strength than in the modulus values (Table 5).

TABLE 5. FIBER PROPERTIES

Type	Tensile Strength (10^3 lb/in.^{2})	Modulus (10^6 lb/in.^{2})	Density (gm/cc)
Thornel 300	373	33.0	1.749
Thornel 75S	308	77.3	1.840
Thornel 75S	324	75.1	1.812
Thornel 75S	364	74.9	1.825

Since each batch of Thornel 75S was sufficient for more than two composite tubes, this variation would not appear in the test results of either the first two full-scale tubes or the smaller test specimens. It may, however, influence the performance of the final tubes.

Small Tube Fabrication

The fabrication of small tubes early in the program was a key factor in the manufacture of reproducible tubes with predictable performance. Processing techniques were continually refined as individual process parameters were sequentially examined during the production of composite tubes 1 in. in diameter and 4.5 in. in length. Seven tubes of this size, suitable for torsional testing, were required before the process was considered to be acceptable. Evaluations of these tubes were in terms of specific gravity, wall thickness, surface condition, and uniformity.

Six additional small tubes were then fabricated, tabbed, and subjected to torsional shear stress. The results of these tests were used in the design analysis and provided guidance for the predictions.

Impact Investigation

The original concept for a low-velocity impact energy absorption system combined a low-density aluminum honeycomb and an outer protective skin. Candidate materials for the skin were commercial Mylar, PVC, and fluorinated ethylene propylene (FEP).

Mylar offers the greatest weight advantage of these materials and provides solvent resistance superior to that of PVC. Although initial evaluations of short sample tube lengths were quite satisfactory, the timely delivery of adequate quantitites of Mylar to complete the program became a serious problem and eventually precluded its use in full-scale production. Consequently, PVC tubing, which was readily available, was examined further and then included in the program.

During the fabrication phase, a relative-type low-velocity impact test was developed using 5-in.-by-5-in. specimens of various constructions. This test showed the effect of a 12-in.-lb impact using a 2.5-in.-diameter steel ball dropped from a height of 5.19 feet. In all cases the impacted specimens rested on an aluminum frame with a 1/2-in. peripheral border that supported the specimens 1/2 in. above a concrete pad. Major damage was sustained by aluminum and unprotected graphite composites. The graphite/epoxy laminate tested was six ply and duplicated the orientation used in the tube construction.

A more detailed analysis of the effects of a low-velocity 12-in.-lb impact was performed, where various combinations of protective coverings were rated according to the percentage of crushing they sustained. Qualitative results showed that at the 70.5-percent level, no physical degradation of a protected composite panel was evident when a 15-power microscopic inspection was conducted on the laminate cross section below the area of impact. A combination of 0.2-in. Flex-Core, two PVC skins, and one foam-filled Flex-Core cell was selected for the final design. Double core and foam also provided adequate protection, but with a weight penalty and diameter increases that exceeded the required envelope dimensions.

Interestingly, in the last test reported a much denser foam was used that crushed the least of any other combination of coverings but did not completely protect the composite. Apparently the added rigidity of the heavier foam permitted transmission of loads sufficient to delaminate the test panel. More definitive tests will be required, including actual tests of mechanical properties after impact, if the optimum trade-off of weight for protection is to be considered.

Final Fabrication

The specific operations required to produce a drive shaft assembly complete with low-velocity impact protective covering are detailed in Whittaker Corporation Research and Development Process Specification No. PS-2021, Appendix B to Reference 2.

PROOF TESTING

Primary emphasis was placed on demonstrating that the drive shaft assemblies could meet the requirements of critical speed and torsional strength. The first two full-scale tubes did exceed these requirements and ultimately were tested to failure at levels [illegible] close to predicted values.

Dynamic Tests

Critical speed tests were conducted on four drive shaft assemblies. A strain-gage displacement transducer with a probe in contact with the outside surface of the tube was used to measure deflection as the speed was continuously increased to over 8000 rpm. The initial test was conducted on a bare composite tube. Displacement, recorded on an X-Y recorder, showed several minor excursions which rapidly dampened, but no indication of sustained oscillation or harmonics was detected. Furthermore, much of the graphic record was generated from vibrations of the motor and table feeding back into the transducer holder. The second tube was spun with virtually identical results. This tube was then completed by adding a protective cover consisting of aluminum Flex-Core and one outer skin of PVC tubing. It was then spun smoothly through the full speed range of 0 - 8000 rpm. However, probe contract was sufficient at the high speeds to cut through the PVC covering.

The final spin test was performed on one of the five units to be delivered to ATL. This assembly included the full protective covering with two PVC skins covering the foam-filled Flex-Core. Again, no critical speeds were detected as the tube was rotated to speeds exceeding 8000 rpm. Visual inspection after the critical speed verified that no physical changes or damages occurred.

Torsion Tests

Following the critical speed tests, the first two composite tubes were strain-gaged and tested to failure in torsion. Ultimate strengths of 15,400 and 15,932 in.-lb were recorded at maximum torsion strains of 0.0105 and 0.0113 in./in. for tubes 1 and 2, respectively. The load was applied hydraulically as a couple, with a loading rate of 3250 in.-lb per minute. Torsional load versus strain was very nearly linear.

Prior to shipment, five fully completed assemblies were subjected to torsion loads exceeding 5000 in.-lb. All units successfully passed this proof test.

X-Ray Inspection

Earlier, each graphite/epoxy tube had been inspected visually, dimensionally checked, tested for specific gravity, and then subjected to X-ray inspection. A section of one of the fractured tubes was used to establish the optimum voltage, film type, and time required to maximize flaw detection sensitivity. Examination of the full-scale X-ray films showed that minute anomalies could be detected. Ply definition was sharp, and it was possible to measure very small gaps in the internal $\pm 45^{\circ}$ layers. No significant difference was noted between tubes, nor was any defect visible.

DESIGN – SANDWICH WALL

Materials Selection

The conceptual drive shaft design consisted of a syntactic epoxy core sandwiched between hybrid composite face sheets. Hysol's ADX-712.2 was the syntactic epoxy selected for the core material. This film, a low-density modification of Hysol's EA 9703 prepreg, was chosen because it is the lightest weight syntactic epoxy prepreg currently available. Typical mechanical and physical properties of this material are given in Table 6.

TABLE 6. TYPICAL PROPERTIES OF ADX-712.2 SYNTACTIC EPOXY

Wet density	0.0224 lb/in.^{3}
Fire retardancy	Yes
Compressive strength	3640 psi
Shear strength	1560 psi
Cured density	0.0185 lb/in.^{3}

Thornel 300 (an intermediate-strength graphite filament) and Kevlar 49 (an organic filament) were selected as reinforcing fibers for the hybrid face sheets. These filaments were selected on the basis of moduli, strength, density, and impact resistance.

To eliminate resin incompatibility, the same matrix was used for the Thornel 300 and Kevlar 49. Unidirectional Thornel 300 tape, unidirectional Kevlar 49 tape, and 120-style Kevlar 49 fabric were purchased, preimpregnated with E-702 epoxy. The E-702 epoxy has a cure temperature of 260°F and a cure cycle compatible with that used for the syntactic epoxy.

Materials characterization tests were conducted on Thornel 300 and Kevlar 49. Five Thornel 300/E-702 panels and five Kevlar 49/E-702 panels were fabricated. Coupons were machined from these panels.

Test data from unidirectional tension coupons and ± 45° angle ply tension coupons were used to compute the in-plane shear stress-strain behavior of unidirectional laminates. In-plane shear stress-strain curves were constructed for both materials.

Design allowables were established for unidirectional Thornel 300/E-702 and unidirectional Kevlar 49/E-702 based on test results. Design allowables for 120-style Kevlar 49 fabric were based on test data developed by the Lockheed-California Company under independent research and development funding.

Design Development Tests

The conceptual drive shaft design assembly fabrication process allows the metallic end fittings to be bonded to the composite shaft face sheets simultaneously with curing of the composite. FM 137U film adhesive was selected for this purpose. Lap shear specimens were fabricated and tested to determine the compatibility of FM 137U adhesive with the E-702 resin used as the matrix material for the composite face sheets. Specimens were cocured using the standard cure cycle recommended for E-702 composites. Test results for these specimens indicated an incompatibility between FM 137U adhesive and the E-702 resin system. All of the specimens failed at the interface between the FM 137U adhesive and the E-702 resin. Inspection of the failed specimens indicated that the FM 137U gelled during the dwell portion of the cure cycle, prior to gelation of the E-702 resin. To alleviate this sequence of events, the cure cycle was modified by reducing the dwell time.

Several double lap shear specimens were fabricated using the revised cure cycle. In addition to FM 137U, 9602.3U and P717U adhesives were tested. Based on the results of that testing, P717U, which withstood the greatest shear stress, was selected for use in the shaft assemblies.

Analysis Methods

The analysis methods used to predict the strength and stiffness of hybrid composite laminates, the torsional stability of orthotropic cylinders, and the critical speed of a rotating shaft are briefly described below.

Computer programs developed at the Lockheed-California Company under independent research and development funding were used for analysis of the shaft designs. The program for hybrid laminate strength analysis, HYBRID, was used extensively. That program calculates the strength and failure steps of hybrid composite laminates under uniaxial or combined loading. Inputs to the program include tension and compression stress-strain behavior of a lamina in the orthotropic axis, in-plane shear stress-strain behavior, and the coefficients of thermal expansion for each material within the laminate. Orientation and thickness of each lamina within the laminate are also input.

The Lockheed computer program COMAIN was used for the torsional stability analysis of the shaft. This program consists of several coupled routines coded for determining the stability of various composite structural elements.

The first mode critical speed, in revolutions per minute, for a uniform, circular shaft with pinned supports at each end was calculated with the following equation from Reference 6:

$$N_{cr} = 30\pi \left(\frac{EX(I)g}{\rho A L^4} \right)^{1/2}$$

[6]Den Hartog, J. P., *Mechanical Vibrations*, New York, McGraw-Hill, 1947.

where EX = axial modulus of elasticity, in.

I = moment of inertia, in.

g = 386 in./sec^2

ρ = density, lb/in.^{3}

A = cross sectional area, in.^{2}

L = shaft length between supports, in.

This equation was used to predict the critical speed of the shaft assembly. Rotational inertia of the end fittings and local changes in stiffness in the vicinity of the end fittings were assumbed to be negligible.

Preliminary Design

The drive shaft design investigated was a sandwich-wall construction containing four materials. Each material performs a distinct function to satisfy the design criteria:

Kevlar 49/E-702 120-style fabric – provides circumferential flexural stiffness required for torsional stability

$\pm \phi$ plies of Kevlar 49/E-702 unidirectional tape – provides torsional strength and stiffness

$\pm \phi$ plies of Thornel 300/E-702 unidirectional tape – provides axial stiffness

Syntactic epoxy core material – separates face sheets for increased circumferential flexural stiffness and strength

The first aspect of the design problem considered was the orientation angle of the Thornel 300/E-702 and Kevlar 49/E-702 laminae. In selecting an orientation angle for the Kevlar 49, a logical choice would be $\pm 45^o$ since this orientation gives maximum shear strength in keeping with the function performed by the Kevlar 49 as noted above. However, a close inspection indicates that the slope of the axial modulus curve for Kevlar 49 increases rapidly from $\pm 45^o$ to $\pm 30^o$. The slope of the shear strength curve does not change as rapidly between $\pm 45^o$ and $\pm 30^o$. Thus, the axial modulus of a $\pm 30^o$ laminate is almost three times that of a $\pm 45^o$ laminate, while the shear strength of the $\pm 45^o$ laminate is only 16 percent greater than that of a $\pm 30^o$ laminate. There fore, an orientation angle of $\pm 30^o$ was selected for the Kevlar 49/E-702 to give the best combination of axial stiffness and shear strength.

The effect of shaft diameter on shear stress, torsional stability, and critical speed was investigated for several laminate configurations. For these analyses, a core thickness of 0.030 in. (minimum available syntactic epoxy) was used. Results of this investigation indicated that the minimum-weight/maximum-length shaft design is obtained by selection of the maximum diameter. Therefore, the shaft outside diameter selected was 4.25 in.

To determine the optimum orientation for the graphite/epoxy laminae, three laminate configurations were investigated for the shaft wall.

For a critical speed of 8000 rpm, the relationship of shaft length to orientation angle of the graphite/epoxy for the three laminate configurations was investigated (See Figure 9). The shaft containing the greatest amount of Thornel 300 has the greatest length. However, selection of the amount and orientation angle of the graphite/epoxy must account for shear strength and shaft weight as well as length.

Shaft strength was predicted for various laminates using the laminate strength analysis program HYBRID. The predicted shear flow at first lamina failure and at laminate rupture versus the orientation angle of the Thornel 300/E-702 for each of the three laminate configurations was investigated.

The laminate which allows the longest minimum-weight shaft must be selected from the laminates satisfying the strength requirements. This was determined by plotting drive shaft system weight per shaft length versus the graphite/epoxy orientation angle for the laminate families being investigated. The system weight includes 15 lb per shaft for end fittings, bearings, and bearing support structure. Thus, the shaft weight parameter is defined as follows:

$$\frac{\text{Shaft Weight}}{\text{Shaft Length}} = \frac{\text{Shaft Weight + Bearing Weight + End Fitting Weight}}{\text{Shaft Length}}$$

This method of analysis was selected because it emphasizes the importance of shaft length on system weight.

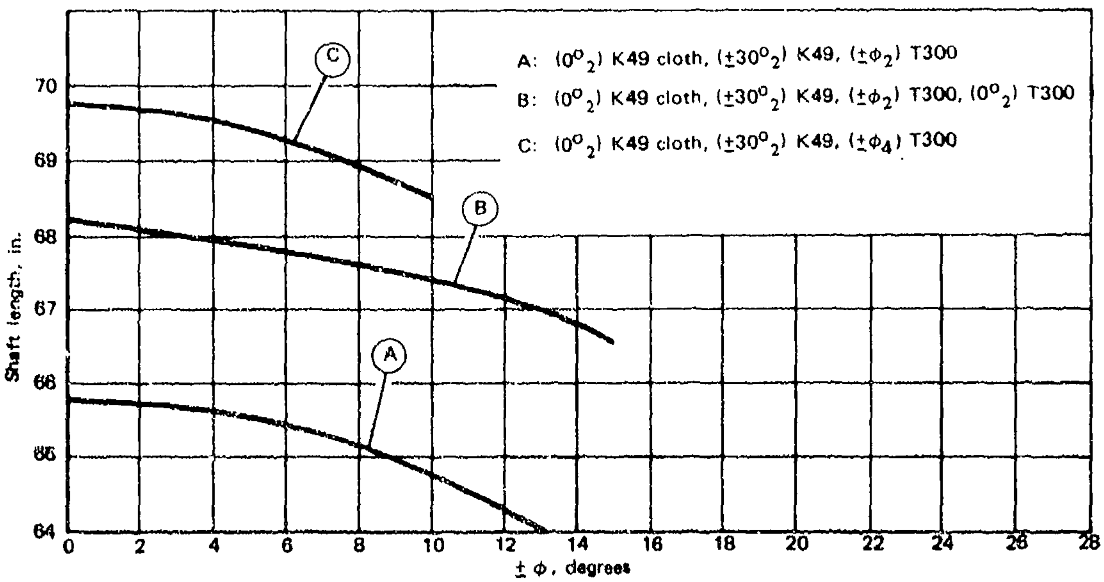


Figure 9. Shaft length versus graphite/epoxy orientation angle for three laminate families.

Final Design

Overall length of the drive shaft is 67 in. with an outside diameter of 4.25 in. The shaft wall has a total thickness of 0.098 in. and consists of the following: two plies of Kevlar 49 120-style cloth, four plies of unidirectional Kevlar 49 tape oriented at $\pm 30^{o}$, four plies of unidirectional Thornel 300 tape oriented at $\pm 10^{o}$, two plies of undirectional Thornel 300 tape oriented at 0^{o}, and 0.030 in. (one ply) of syntactic epoxy. Integral end fittings were included in the shaft design. Titanium was selected for these fittings because it has a high specific shear strength and a coefficient of thermal expansion compatible with the hybrid laminate. The primary advantages of the integral end fittings include the use of a double lap shear joint to reduce eccentricities and lap length, and a simplification of the fabrication process.

A Kevlar 49 doubler was placed on the interior of the shaft in the transition region from the shaft to the end fittings to facilitate the load transition from the end fitting to the shaft wall and to eliminate the possibility of premature failure at the shaft end fitting.

FABRICATION

Eight composite shaft assemblies were built during the program using a tooling concept and manufacturing method developed by the Lockheed-California Company under independent research and development funds. The basic tooling concept was to apply laminating pressure to the inside surface of the cylindrical layup and to react it with a female tool. The tooling system permits the removal of the layup mandrel from the interior of the cured composite tube without disassembling it. The reusable mandrel was easily removed from the shaft interior even with the internal end doublers in place. This tooling technique allowed the cocuring of the basic composite shaft and the bonding of the composite cylinder to the end fittings.

Due to the limited quantity of prototype shafts built, automated fabrication techniques such as use of a tape laying machine were not employed. The shafts were laid up by hand using 3-in.-wide preimpregnated tape. The laid-up shaft assembly was placed in the female tool and cured in the autoclave.

The measured weights for the eight shaft assemblies are presented in Table 7.

Additional resin was added to the exterior and interior plies of 120-style fabric on shaft assemblies 6, 7, and 8 to overcome a resin-starved surface condition that had occurred in spots on previous shafts. This additional resin is responsible for the weight differential between shafts 6, 7, and 8 and the previous shafts. It was concluded that the 120-style fabric should have been purchased with a greater resin content.

TABLE 7. MEASURED SHAFT ASSEMBLY WEIGHTS

Shaft Number	Weight (lb)
1	4.69
2	*
3	4.73
4	4.70
5	4.81
6	5.00
7	5.12
8	4.81

*Rejected due to autoclave malfunction during cure cycle.

PROOF TESTING

Dynamic Tests

Shaft assemblies 1 and 3 were dynamically tested to determine their first bending mode frequencies for a simply supported (hinge-hinge) end condition. The first bending mode of each shaft was determined by swept frequency excitation of the shaft applied by an electrodynamic shaker through a flexure element attached at one end of the shaft. The opposite end of the shaft was hinge-supported to ground.

The test procedure used for the dynamic tests was to "tune in" with a low input excitation level for maximum excursion at a single anti-node point at the center of the shaft (characteristic of the first bending mode). This "tune-in" frequency was the first bending mode frequency and was read out on a digital electronic frequency counter.

The measured first bending mode frequencies on shafts 1 and 3 were 122.9 Hz and 123.2 Hz, respectively. This corresponds to shaft critical speeds of 7374 and 7392 rpm for shafts 1 and 3, approximately 9.5 percent lower than the predicted frequency of 135.7 Hz (8144 rpm). Two factors were responsible for the discrepancy between predicted and actual frequency. First, the measured shaft weight was greater than the predicted shaft weight used in the dynamic analysis. Secondly, the frequency was predicted assuming a shaft with a uniform bending stiffness distribution. Thus, the analysis ignored the rotational inertia of the end fittings and the local change in specific stiffness at the end fittings.

Static Torsion Tests

Shafts 1 and 3 were statically tested to failure in torsion. The test setup consisted of a hydraulic jack and a loading fixture comprising a torque arm and a reaction plate. Hydraulic pressure to the loading jack was controlled through an Edison load maintainer. All test loads were monitored with a load cell mounted in series in the jack train. Deflections were measured by a deflection transducer. The output of the load cell and deflection transducer was recorded by the Central Data System (CDS). Data points were recorded during the continuous loading phase at a rate of 30 per second.

Test loads were applied in two runs. The loading during the first run was terminated abruptly at the point where failure appeared to be imminent. The loading during the second run was continuous until a definite failure was produced in the specimen.

On the initial loading, shaft 1 was loaded to 17,300 in.-lb of torque when instability began to occur. The load was dropped and the shaft reloaded. At a torque of 15,500 in.-lb, general instability occurred followed by rupture. The predicted rupture strength of the shaft was 26,900 in.-lb and the predicted torsional stability strength was 28,500 in.-lb.

The shear stress-strain response of the shaft hybrid laminate was predicted analytically. Lamina failures and subsequent changes in laminate stiffness were also identified analytically. Shaft instability occurred at a stress level approximately the same as the predicted stress level for compression failure of the 30° Kevlar 49 plies.

An analysis was conducted to determine the torsional stability of the shaft, assuming that the + 30° plies of Kevlar 49 had failed in compression. This analysis predicted torsional instability of the shaft at a torque of 25,000 in.-lb. Thus, it appears that the shaft rupture was initiated by a torsional instability precipitated by failure at the 30° plies of Kevlar 49 in compression.

Shaft 3 was tested in torsion in an identical manner to shaft 1. The maximum torque on the first load application was 18,000 in.-lb. On the second loading, the maximum torque was 17,000 in.-lb. The failure mode of shaft 3 was identical with that of shaft 1.

To determine the rupture strength of the shaft, two short shaft specimens were assembled from segments of shafts 1 and 3. The short shaft specimens were approximately 25 in. long, with the standard titanium fitting on one end and an aluminum fitting attached by fiberglass doublers on the other end.

Short shaft 1 (taken from shaft assembly 1) was instrumented with three strain gages oriented at + 45°, 0°, and - 45° to the shaft longitudinal axis. The specimen failed at a torque of 30,400 in.-lb.

Short shaft 2 (taken from shaft assembly 3) was tested to determine the residual strength of the shaft after ballistic damage. To simulate the damage caused by a tumbling .30-caliber projectile, a 1.5-in. long slot was cut on one side of the shaft. The specimen was loaded to 11,000 in.-lb and inspected. No crack growth was observed. A second 1.5-in.-long slot was cut in the shaft at 180° from the first slot. The specimen was loaded to failure, which occurred at a torque of 14,000 in.-lb. Thus, the ratio of undamaged torque to damaged torque for the simulated fully tumbled .30-caliber projectile was 2.17.

Proof tests were conducted on shaft assemblies 4 through 8. Each shaft was loaded with a torque of 5000 in.-lb. A load-deflection curve was plotted for each shaft. Comparison of these curves indicated negligible variations in torsional stiffness in the shaft assemblies.

Impact and Ballistic Tests

Shaft 2, the rejected shaft, was tested for impact resistance using a ball-drop test. Impacts of 12 ft-lb and 16.1 ft-lb produced no visible damage. The shaft was sectioned in

the vicinity of the 16.1 ft-lb impact. The syntactic epoxy had cracked locally and there was a localized decrease in the flexural stiffness of the shaft wall.

An impact of 23.7 ft-lb caused visible damage to the shaft in the form of localized crazing. It appeared that the graphite/epoxy plies cracked transversely to the fiber direction. Tap testing indicated delaminations in the vicinity of the impact; however, it appeared that the ±30° Kevlar 49 was intact. Thus, the shaft should still have a great portion of its load-carrying capability.

Shaft 2 was ballistically impacted with a .22-caliber (long rifle) projectile fired at close range and 0° obliquity. Prior to the test, the outer ply of 120-style fabric Kevlar 49 was removed. Entry holes in both shaft walls were very clean. The exit hole on the inside of the shaft where the 120-style cloth remained was also clean, while the exit hole on the exterior of the shaft was quite ragged. Portions of the outer laminate tended to peel away from the hole with the resulting damage to the laminate being much greater than that observed on the exit hole in the interior of the shaft. These observations indicate the importance of the 120-style cloth in minimizing ballistic damage.

DESIGN – FILAMENT-WOUND

The design of the drive shaft was based on a fabrication process of wet filament winding over an air-inflated plastic mandrel that is placed in a female mold while the resin is being cured. This process allows the shaft end fittings to be made an integral part of the shaft and is amenable to producing a straight and concentric shaft of high quality. However, in order to take advantage of the mold during curing of the resin, circumferential (hoop) windings could not be used since they would restrict the radial growth of the composite and would not allow the windings to contact the mold.

Since the damage incurred by bullet impact could not be accurately assessed, the shaft was designed to include a sizeable safety factor while meeting the criteria for minimum length and critical speed.

The fibers and resin (Thornel 300/epoxy) were selected on the basis of strength and stiffness requirements and prior experience. The characteristics of the Thornel 300 and epoxy resin are summarized in Table 8.

TABLE 8. RAW MATERIAL PROPERTIES

Property	Thornel 300	Epoxy*
E_x, 10^6 psi	34.0	0.47
E_y, 10^6 psi	1.3	0.47
G_{xy}, 10^6 psi	3.5	0.17
F_{tu}, psi	240,000	10,000
F_{cu}, psi	200,000	20,000
ρ, lb/in.^{3}	0.0636	0.0412

*APCO 24/34APCO 2435, Applied Plastics Co.

FABRICATION

The drive shafts were fabricated in accordance with FSI Process Specification 504-201, Appendix III to Reference 4.

The winding was a three-circuit geodesic pattern using six Thornel 300 rovings to make a band 0.204 inch wide.

The principal problems encountered during fabrication concerned development of:

1. A machine setup and delivery system that would place the fibers along a geodesic path in the fitting necked-down area

2. A delivery system to handle the graphite rovings without breaking them

3. A process for molding the outside surface at the shaft during curing

4. A process for removing the plastic liner from the finished shaft

The first three problems were solved satisfactorily during the fabrication process. The plastic liner was left inside the finished shafts delivered to ATL.

PROOF TESTING

Three drive shafts were tested at FSI. Two shafts were subjected to spin testing to 8000 rpm and then torque tested to failure. The third shaft was subjected to torque testing to failure only.

The test requirements were to determine that there were no critical speeds below 8000 rpm and that the ultimate torsional strength of the basic tube was above 11,000 in.-lb.

The shafts were inspected before testing. Their dimensions are given in Table 9.

TABLE 9. SHAFT DIMENSIONS

Shaft No.	Total Length (in.)	Weight (lb)	Shaft O.D. (in.)	Wall Thickness (in.)
1	58.25	5.3	4.190 - 4.220	0.108 - 0.123
2	58.18	4.75	4.196 - 4.210	0.092 - 0.108
3	58.31	6.3*	4.226 - 4.223	0.112 - 0.115

*With end reinforcement material

Dynamic Tests

The shafts were dynamically balanced prior to testing, using a Stewart-Warner Model 704 resonance balancer modified to accommodate drive shafts. Balancing weights were bonded to the shafts as required to obtain an adequate dynamic balance.

Testing was accomplished using a spin test fixture. The shafts were installed in the fixture and rotated to a minimum of 8000 rpm. An electronic tachometer connected to a magnetic pickup mounted on the test fixture was used to indicate the rpm.

Shafts 1 and 2 exhibited no critical resonances and sustained no damage from the dynamic tests. Shaft 3 was not dynamically tested.

Torsion Tests

The shafts were installed in a torque test fixture. The fixture allowed the shaft to rotate without resistance around its centerline. One end of the shaft was restrained from rotation by an 8-in. arm attached to a dynamometer. Force was applied to the opposite end by a hydraulic jack through another 8-in. arm. Torque was calculated by multiplying the reading from the dynamometer by the length of the arm.

Shaft 1 failed at 18,000 in.-lb torque. Failure occurred in the end fitting. Shaft 2 failed at 16,800 in.-lb, also in an end fitting. The ends of shaft 2 were reinforced and the shaft was retested. Failure occurred the second time in the tube portion at 24,000 in.-lb.

Shaft 3 failed at 27,200 in.-lb in a slightly reinforced end fitting. The ends were reinforced similarly to those of shaft 2 and the shaft was retested. Failure occurred the second time at 30,400 in.-lb in the tube portion.

GOVERNMENT EVALUATION TESTING

For low energy impact testing as a part of the Government evaluation, ATL conducted drop testing with a 3-in.-diameter steel ball weighing 4 pounds at energy drop levels up to 20 ft-lb in 4-ft-lb increments. Following each drop, the shaft was visually inspected using a borescope for internal inspection. The shafts were then fatigue tested on the torsional fatigue machine for 20,000 cycles at 4600 ± 550 in.-lb.

A nondestructive test (NDT) inspection was then performed followed by ultimate testing.

COMPOSITE DRIVE SHAFT TEST PROCEDURE

Five identical specimens (described in Table 10) from each contractor were received and subjected to the following tests:

Specimen 1

1-1 Statically test on torsion machine to 11,000 in.-lb, measuring deflection

1-2 Fatigue test in torsion, 4600 in.-lb ± 550 alternating, 10^6 cycles or to failure

1-3 Repeat 1-1

1-4 Fatigue test in torsion, 4600 in.-lb ± 550 alternating, 500,000 cycles or to failure

1-5 Repeat 1-1

1-6 Fatigue test in torsion, 9200 ± 1100 alternating, 500,000 cycles or to failure

1-7 Repeat 1-1

Specimen 2

2-1 Statically test on torsion machine to 11,000 in.-lb, measuring deflection

2-2 Ballistically impact with tumbled .30-caliber round at 2700 fps

2-3 Statically test to 7500 in.-lb, measuring deflection

2-4 Fatigue test in torsion, 4600 in.-lb ± 550 alternating, 2000 cycles or to failure

2-5 Repeat 2-3

2-6 Fatigue test in torsion, 6900 in.-lb ± 550 alternating, 2000 cycles or to failure

2-7 Repeat 2-3

2-8 If no failure has occurred, impact with another round, repeating 2-2 to 2-7

Specimen 3

3-1 Statically test on torsion machine to 11,000 in.-lb, measuring deflection

3-2 Ball-drop test

3-3 NDT of impact area

3-4 Statically test on torsion machine to 7500 in.-lb, measuring deflection

3-5 Fatigue in torsion 4600 in.-lb ± 550 alternating, 40,000 cycles or to failure

3-6 Repeat 3-4

3-7 Fatigue in torsion 6900 in.-lb ± 550 alternating, 40,000 cycles or to failure

3-8 Repeat 3-4

3-9 If no failure has occurred, perform another ball-drop test at increased impact energy, repeating 3-2 to 3-8

Specimen 4

Ballistically test specimen with untumbled .50-caliber round at 2900 fps. Procedure will be identical to Specimen 2, with change in caliber.

Specimen 5

Used as a spare for Specimens 1-- 4.

TEST RESULTS

Test results are shown in Table 11.

TABLE 10. COMPOSITE SHAFT CHARACTERISTICS

	WHITTAKER CORP.	LOCKHEED-CALIFORNIA CO.	FIBER SCIENCE, INC.
Construction	Graphite/epoxy with impact-resistant cover	Graphite/Kevlar/epoxy/syntactic foam sandwich	Graphite/epoxy filament wound
Length, in.	56.5	67.0	58.2
Wall Thickness, in.	.038 (including cover)	0.098 (including 0.030-in. foam	0.110
Weight, lb	3.0	4.8	5.0 (plus 1-lb end reinforcement)
Weight, Uniform Section, lb/ft	0.434	0.75	0.92

TABLE 11. TEST RESULTS

COMPOSITE SHAFTING TEST RESULTS

	Fatigue test only*	• .30-cal fully tumbled round (center hit), 2700 fps, proof load • Fatigue test (4 x 10^3 cycles)	• .30-cal fully tumbled round (edge hit), 2700 fps, proof load • Fatigue test (4 x 10^3 cycles)	• .50-cal impact, 2900 fps, proof load • Fatigue test (4 x 10^3 cycles)	• Ball drop, proof load • Fatigue test (2 x 10^4 cycles)
Whittaker	Failed in ultimate test at 15,400 in.-lb	Failed at 5300 in.-lb		Sustained single impact. Failed at 6700 in.-lb	Failed after four 8-ft-lb ball drops at 5400 in.-lb
Lockheed	Failed in ultimate test at 19,700 in.-lb	No failure	Failed at 3400 in.-lb	Sustained two impacts. Failed at 19,000 in.-lb	Successful after 4-, 8-, 12-, 16-, and 20-ft-lb drop test. Failed in ultimate test at 18,900 in.-lb
Fiber Science	Failed in ultimate test at 31,500 in.-lb	No failure	Failed at 3400 in.-lb	Sustained two impacts. Failed in ultimate test at 24,000 in.-lb	Successful after 4-, 8-, 12-, 16-, and 20-ft-lb drop test. Failed in ultimate test at 22,500 in.-lb

*Fatigue Test

1 x 10^6 cycles	4600 ± 550 in.-lb
5 x 10^5 cycles	4600 ± 1100 in.-lb
5 x 10^5 cycles	9200 ± 1100 in.-lb

CONCLUSIONS

1. Based on tests conducted by contractors and by ATL, both the sandwich-wall and the wet-filament-wound drive shafts met or exceeded the stated design requirements. The graphite tape-wound approach with the protective cover was incapable of meeting the .30-caliber, .50-caliber, and ball-drop tests.

2. The majority of test failures occurred in shaft end fittings.

3. Uniform and reproducible composite drive shafts of predictably high quality can be produced that meet U.S. Army dynamic, structural, and low-energy damage resistance requirements.

4. Tooling and fabrication concepts were demonstrated to be satisfactory.

5. In-service evaluation of composite tail rotor drive shafts is desirable.

RECOMMENDATIONS

1. Recommend further programs be initiated to refine drive shaft design and fabrication concepts developed under the project covered by this report. Additional research should be particularly directed toward improved shaft end fitting design.

2. Recommend field testing of composite tail rotor drive shafts.